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Lewis Research Center
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SUMMARY

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Residual stress measurements were made in the zone of resolved shearing stress on five SAE 52100 upper test ball specimens with an average Rockwell C hardness of 63.2. These upper test ball specimens were run against lower test balls of nominal Rockwell C hardnesses of 60, 62, 63, 65, and 66 until either or both components failed because of fatigue. Test conditions included an average race temperature of 150° F, a maximum (Hertz) compressive stress of 800,000 psi, and a highly purified naphthenic mineral oil lubricant. Fatigue lives of the upper test balls were compared with measured residual stresses in the subsurface zone of resolved shearing stress.

The compressive residual stress induced in the upper test ball during running is a function of ΔH, the hardness of the lower test balls minus the upper test ball hardness. An interrelation is indicated among differences in component hardness, induced compressive residual stress, and fatigue life. Measured values of compressive residual stress within the zone of resolved maximum shearing stress ranged from 178,000 and 294,000 psi. The apparent maximum residual stress occurs above where $\Delta H = 0$.

AUTHOR

INTRODUCTION

Much research has been directed towards increasing the fatigue life of ball and roller bearings and gears and, hence, their reliability. These efforts have led to increased operational reliability in engine and other aerospace equipment and components (ref. 1). Rolling-contact fatigue research reported in reference 2 indicated that, in the NASA five-ball fatigue tester, system fatigue life and load capacity were maximum where the lower test ball hardness was one to two points (Rockwell C) greater than the upper test ball hardness for varying hardnesses of both components. Differences in plastic deformation and contact temperature for different hardness combinations could not account for measured differences in fatigue life. These results indicate that a maximum bearing fatigue life and reliability can be achieved where the balls of the bearing are one to two points (Rockwell C) harder than the races.

It was shown in references 3 and 4 that residual compressive stresses were developed below rolling-contact surfaces, the magnitude of which appeared to be a function of time. Additionally, residual compressive stresses induced by mechanical processing operations were found to increase the fatigue life of balls and complete bearings (ref. 5). A unit volume on the upper ball is stressed many more times than a point on any of the lower test balls, so that it would be likely that the upper test ball would absorb more energy and build up a proportionally greater amount of subsurface residual stress than in each of the lower balls. If these residual stresses were compressive, they would act to reduce the maximum shearing stress, which is believed to be of prime importance in rolling-contact fatigue (see appendix A).

The research reported herein was undertaken to determine (1) if residual stresses were induced in the subsurface zone of resolved maximum shearing stress and (2) if the induced residual stresses correlate with component hardness combinations and fatigue life. Residual stress measurements were performed on upper test balls having an average Rockwell C hardness of 63.2 run against lower test balls having Rockwell C hardnesses of 59.7 to 66.4. The test balls were run in the five-ball fatigue tester at a maximum initial Hertz stress of 800,000 psi, a shaft speed of 10,000 rpm, a contact angle of 30°, and a nominal temperature of 150° F with a highly refined napthenic mineral oil as the lubricant. All results were obtained with a single lubricant and batch of material for the upper test balls.

APPARATUS AND PROCEDURE

Test and Related Equipment

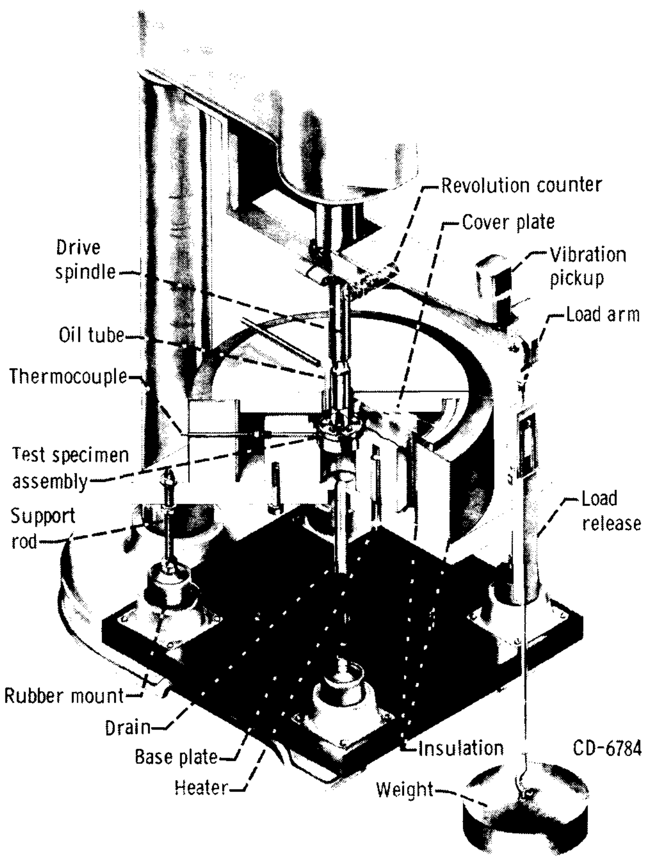
The NASA five-ball fatigue tester shown in figure 1 was used to stress the 1/2-inch-diameter SAE 52100 steel upper test balls, tempered to an average Rockwell C hardness of 63.2 and run against groups of lower test balls of the same material of varying hardnesses. Essentially this fatigue apparatus consists of a 1/2-inch-diameter test ball pyramided upon four 1/2-inch-diameter lower test balls that are positioned by a separator and are free to rotate in an angular contact raceway (see fig. 1(b)). A more detailed description of this apparatus is given in reference 6.

The upper test ball is analogous in operation to the inner race of a ball bearing, while the lower test balls and the angular contact raceway are analogous to the balls and the outer race of a ball bearing, respectively. For every revolution of the drive shaft, the upper test ball receives 3 stress cycles.

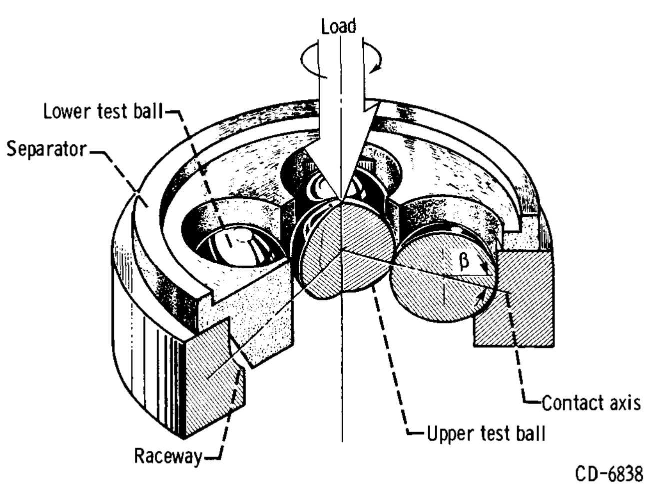
Standard X-ray diffraction and microscopy equipment were used to obtain the results reported herein.

Procedure

Five upper test balls having a Rockwell C hardness of 63.2 were run against support balls having Rockwell C hardnesses of 59.7, 61.8, 63.4, 65.0, and 66.2



(a) Cutaway view of five-ball fatigue tester.



(b) Schematic of five-ball fatigue tester.
Figure 1. - Test apparatus.

at a contact angle of 30°. These combinations and the number of stress cycles to which they were run are given in table I. The various systems were run from 32 to 40 million stress cycles and failed by fatigue.

The upper test balls were electropolished to a depth of 0.005 inch over a third of the ball surface area with a part of the track in the middle of the electropolished area. The geometry of the surface was well preserved during electropolishing, and the running track was still visible.

Standard X-ray diffraction techniques were used to measure residual stresses (ref. 7). The measurements for residual stresses in these balls were made by Mr. Ragnar Lindgren and Dr. W. E. Littmann of The Timken Roller Bearing Company, Canton, Ohio.

RESULTS AND DISCUSSION

Effect of Component Hardness on Fatigue Life

In general, research reported in reference 2 indicated that, for a specific upper test ball hardness, rolling-contact life and load-carrying capacity of the test system reached a maximum as a function of ΔH, the lower ball hardness minus the upper ball hardness. The peak life - hardness combination occurred where the hardness of the lower test balls was approximately one to two points (Rockwell C) greater than the upper test ball.

Conditions under which the tests of reference 2 were run were an average race temperature of 150° F due to frictional heating, a maximum Hertz stress of 800,000 psi, and a highly purified naphthenic mineral oil lubricant. The three groups of

TABLE I. - RESIDUAL STRESS MEASUREMENTS OF UPPER TEST BALL SPECIMENS

HAVING ROCKWELL C HARDNESS OF 63.2

Specimen number	Lower test ball Rockwell C hardness	Difference in Rockwell C hardness between lower and upper test balls, ΔH	Specimen running time, millions of stress cycles	Measured residual stress at depth of 0.005 inch below ball surface removed by electropolishing, psi		Calculated 10-percent life of upper test ball based on residual stresses,[a] millions of stress cycles
				Under track	Outside track	
1	59.7	-3.5	36.1	[b]-178×10^3	[b]-59×10^3	0.391
2	61.8	-1.4	32.4	-198	0	.641
3	63.4	.2	37.9	-294	0	11.3
4	65.0	1.8	40.0	-223	0	1.278
5	66.2	3.0	39.1	-257	-20	3.23

[a]Based on experimental 10-percent life of 11.3×10^6 stress cycles at $\Delta H = 0.2$.

[b]Negative sign indicates compressor residual stress.

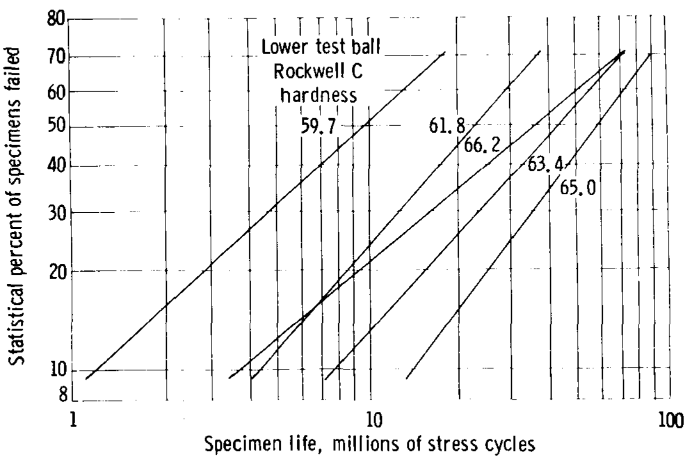


Figure 2. - Summary of rolling-contact fatigue lives of five-ball system composed of SAE 52100 steel lower test balls of varying hardness run with Rockwell C hardness 63.2 SAE 52100 steel upper test ball at a maximum Hertz stress of 800,000 psi. Room temperature; highly purified napthenic mineral oil. (Data taken from ref. 2.)

upper test balls having average Rockwell C hardnesses of 60.5, 63.2, and 65.2 were run against lower test balls having nominal Rockwell C hardnesses of 60, 62, 63, 65, and 66. Both lower and upper test balls were from the same batch of SAE 52100 steel except for the balls with a Rockwell C hardness of 66.

Examination of reference 2 shows that the data obtained with the test balls having an average Rockwell C hardness of 63.2 were generally representative of all other data obtained. These data for the system life with the upper test balls having a Rockwell C hardness of 63.2 are summarized in figure 2 and table II. A plot of system 10-percent life as a function of ΔH is given in figure 3. There is a peak life at a ΔH of approximately 1.8 (Rockwell C).

From these data (ref. 2), the 10-percent lives of the upper and the lower test balls were determined and are summarized in table III. These data are plotted separately as a function of ΔH in figure 3. As for the system life, it can be seen from figure 3 that component lives are also a function of ΔH. The upper test ball life, however, appears to control the trend in system life.

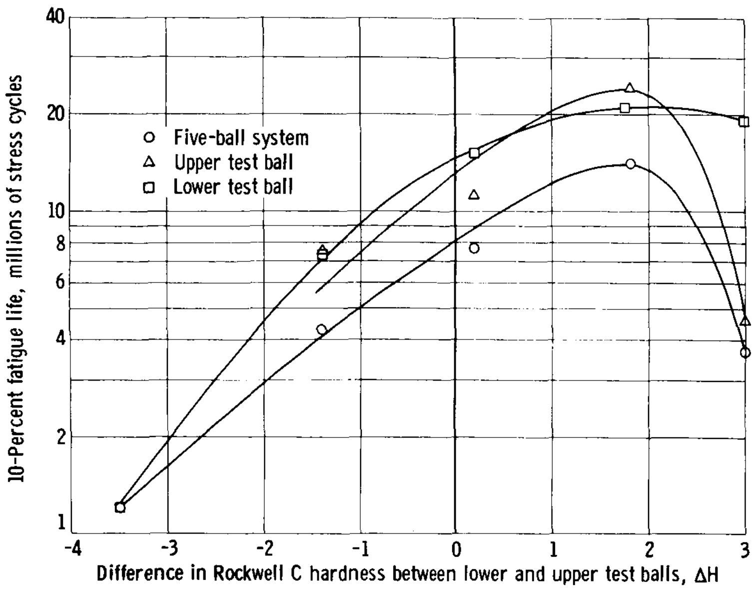


Figure 3. - 10-Percent life of five-ball system and components as function of ΔH for upper test balls having Rockwell C hardness of 63.2.

TABLE II. - STATISTICAL SYSTEM FATIGUE LIVES AND LOAD CAPACITIES OF FIVE-BALL FATIGUE TESTER FOR UPPER TEST BALL SPECIMENS HAVING ROCKWELL C HARDNESS OF 63.2 RUN AGAINST LOWER TEST BALLS OF VARYING HARDNESSES

[Initial maximum Hertz stress, 800,000 psi; contact angle, 30°; material, SAE 52100 steel; room temperature. Data taken from ref. 2.]

Lower test ball Rockwell C hardness	Difference in Rockwell C hardness between lower and upper test balls, ΔH	System thrust load, P, lb	System 10-percent fatigue life, millions of stress cycles	System load capacity based on experimental life, C,[a] lb	Confidence number,[b] percent	Failure index (number of failures out of number of specimens tested)	Number of upper test ball failures	Number of lower test ball failures	Number of upper and lower test ball failures	Percent of upper test ball failures
59.7	-3.5	340	1.2	362	99	20 out of 21	0	20	0	0
61.8	-1.4	↓	4.3	553	92	22 out of 22	6	13	3	27
63.4	.2	↓	7.7	670	76	22 out of 23	13	3	6	59
65.0	1.8	↓	14.2	830	--	22 out of 22	12	4	6	55
66.2	3.0	↓	3.7	525	91	22 out of 23	18	2	2	82

[a] $C = P\sqrt[3]{L}$ where P is load on test system and L is 10-percent life of system.

[b] Percentage of time that 10-percent life obtained with each hardness combination will have same relation to hardness combination in that series exhibiting highest 10-percent life.

TABLE III. - COMPONENT LIVES OF FIVE-BALL FATIGUE TESTER FOR UPPER TEST BALL HAVING ROCKWELL C HARDNESS 63.2 RUN AGAINST LOWER TEST BALLS OF VARYING HARDNESSES

[Initial maximum Hertz stress, 800,000 psi; contact angle, 30°; material, SAE 52100 steel; room temperature. Data taken from ref. 2.]

Lower test ball Rockwell C hardness	Upper test ball		Lower test ball	
	10-Percent life, millions of stress cycles	50-Percent life, millions of stress cycles	10-Percent life, millions of stress cycles	50-Percent life, millions of stress cycles
59.7	----	--	1.2	9.8
61.8	7.4	59	7.5	28.5
63.4	11.3	53	15.0	125
65.0	24.0	85	21.3	97
66.2	4.6	42	19.3	330

Effect of Residual Stress on Maximum Shearing Stress

Induced residual stress can either increase or decrease the maximum shearing stress, according to the following equations:

$$\left(\tau_{max}\right)_r = -3.22\times10^6 \left(\frac{P_N}{R^2 S_{max}}\right)^{1/2} - \frac{1}{2} \left(\pm Sr_y\right) \qquad \text{(A9)}$$

where the positive or negative sign of Sr_y indicates a tensile or a compressive residual stress, respectively. Accordingly, a compressive residual stress would reduce the maximum shearing stress and increase fatigue life according to the inverse relation of life and stress to the ninth power (see appendix B)

$$L \propto \left[\frac{1}{(\tau_{max})_r}\right]^9 \qquad \text{(B4)}$$

Residual Stress Measurements

The resulting residual stress measurements are shown in table I. The measured residual stresses below the track were compressive and varied between 178,000 and 294,000 psi. These values were considerably higher than those anticipated, based on the data from references 3 and 4. Even so, it should be noted that these measured stresses are less than the true values because the X-ray beam could not be focused entirely in the stressed zone. Measurements of stress outside the stressed zone showed background compressive residual stresses in samples 1 and 5 of 59,000 and 20,000 psi, respectively. These background

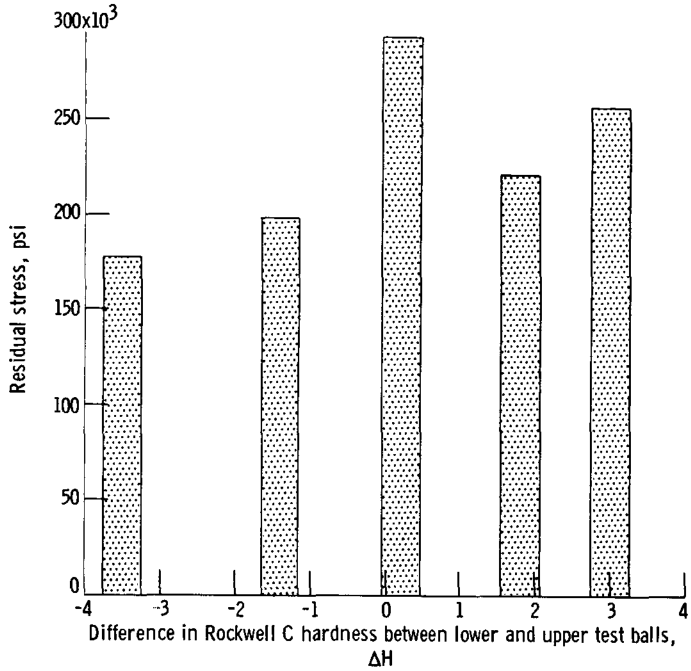


Figure 4. - Measured compressive residual stress in track of upper test balls having Rockwell C hardness of 63.2 as function of ΔH.

stresses would have a tendency to increase the value of the measured residual stresses in the stressed zone. Consequently, the values for samples 1 and 5 might be high relative to the other samples.

The measured compressive residual stresses are plotted as a function of ΔH in figure 4. From this figure, it is noted that the measured stress increases with increasing lower test ball hardness to an intermediate hardness where a peak was obtained. For further increases in lower test ball hardness, the measured residual stress decreases. On the basis of these limited data, the apparent maximum residual stress occurs above where $\Delta H = 0$.

Effect of Measured Compressive Residual Stress on Fatigue Life

The measured values of compressive residual stress were used to calculate the maximum shearing stress (see the section Effect of Residual Stress on Maximum Shearing Stress). Based on the measured values of residual stress and the 10-percent life of the upper test ball at $\Delta H = 0.2$, theoretical absolute lives were calculated using the inverse relation of life and stress to the ninth power (see eq. (B7)) and assuming that the measured residual stress existed throughout each test. These values are summarized in table I and plotted as a function of ΔH in figure 5(a). For comparison purposes, the experimental upper test ball fatigue results are given in figure 5(b). From these data, it can be seen that the calculated and experimental lives are within a reasonable range of each other. The results indicate that an interrelation exists among differences in component hardness, induced compressive residual stress, and fatigue life.

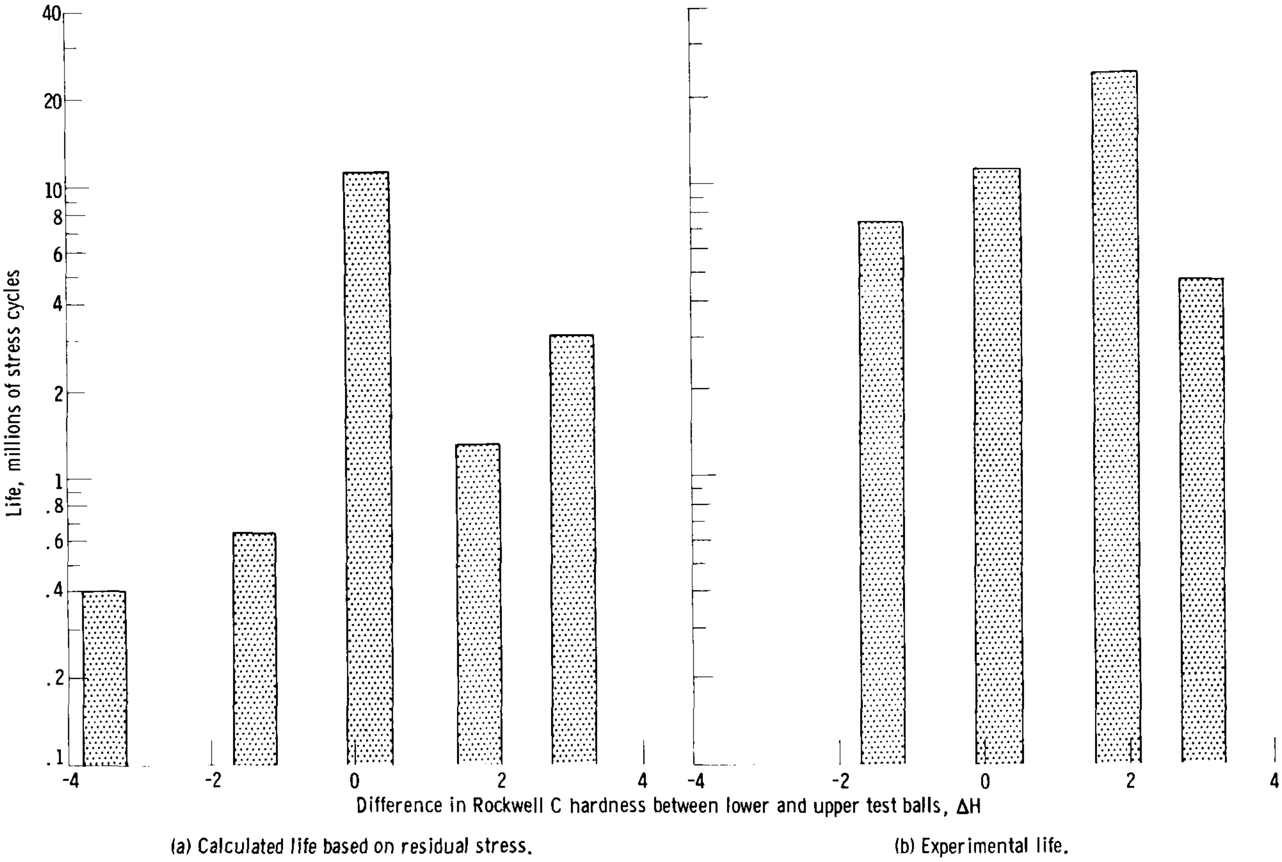


Figure 5. - Calculated and experimental 10-percent lives of upper test balls of Rockwell C hardness 63.2 SAE 52100 steel run against lower test balls of varying hardnesses of the same material as function of ΔH.

SUMMARY OF RESULTS

Five upper test balls having a Rockwell C hardness of 63.2 were run in the NASA five-ball tester against support balls having nominal Rockwell C hardnesses of 60 to 66. Tests were conducted at a maximum Hertz stress of 800,000 psi, no heat added, and with a highly purified naphthenic mineral oil lubricant. The upper test balls were electropolished to a depth of 0.005 inch, and residual stress measurements were made of the upper ball track by means of X-ray techniques. The following results were obtained:

1. The magnitude of the residual stress induced into the upper test ball during running is a function of ΔH, the hardness of the lower test balls minus the upper test ball hardness.

2. An interrelation is indicated among differences in component hardness, induced compressive residual stress, and fatigue life.

3. Measured values of residual stress within the zone of resolved maximum hearing stress ranged from 178,000 to 294,000 psi and were compressive. The apparent maximum residual stress occurs above where $\Delta H = 0$.

Lewis Research Center
National Aeronautics and Space Administration
Cleveland, Ohio, December 2, 1964

APPENDIX A

DERIVATION OF RESIDUAL STRESS EFFECT ON MAXIMUM SHEARING STRESS

It has been shown by investigators (refs. 8 to 10) that a mode of classical rolling-contact fatigue begins within the subsurface zone of resolved shearing stress. It has been further speculated that the stress instrumental in causing fatigue is the maximum shearing stress in this zone (refs. 9 and 10). It therefore becomes necessary to analyze the effect of residual stresses that can be induced in the subsurface zone on the maximum shearing stress.

The maximum shearing stress at any point on a plane of a stress volume τ_{max} is

$$\tau_{max} = \frac{1}{2}(S_{max} - S_{min})$$

where S_{max} and S_{min} are stresses in principal directions at that point. It has been shown that, for a ball rolling in a conforming groove, a theoretical critical distance below the surface exists where

$$\tau_{max} = \frac{1}{2}(S_z - S_y)$$

where S_z is a principal compressive stress in a direction normal to the contact area and S_y is a principal compressive stress parallel to the direction of rolling (ref. 11 and fig. 6).

For the case of a sphere loaded statically against another sphere, the maximum theoretical shearing stress occurs in the x-z and the y-z planes since the values of principal compressive stress in the x and the y directions at any distance below the surface are equal (circular contact area). Therefore,

$$\tau_{max} = \frac{1}{2}(S_z - S_y) = \frac{1}{2}(S_z - S_x) \qquad \text{(A1)}$$

where S_x is a principal compressive stress perpendicular to the direction of rolling. This equation is based upon the assumption that there are no residual stresses present in the material.

If residual stresses of equal value and type are formed in the x and y directions, then, for a circular contact area in the y-z plane,

$$\left(\tau_{max}\right)_r = \frac{1}{2}\left[S_z - (S_y \pm Sr_y)\right] \qquad \text{(A2)}$$

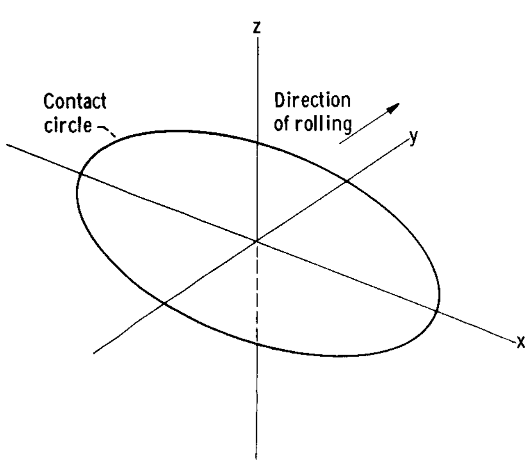


Figure 6. - Coordinate system used for analysis of effect of residual stress on fatigue life.

and one in the x-z plane,

$$\left(\tau_{max}\right)_r = \frac{1}{2}\left[S_z - (S_x \pm Sr_x)\right] \qquad (A3)$$

where $\pm Sr_x$ and $\pm Sr_y$ are residual stresses in the x and y directions, respectively. The plus or minus sign refers to tensile or compressive stress, respectively. From reference 11, for a sphere loaded on a sphere,

$$\tau_{max} = -0.205 \frac{a}{\Lambda} \qquad (A4)$$

where a is the radius of the circular contact area,

$$a = \left(\frac{3}{2\pi}\frac{P_N}{S_{max}}\right)^{1/2} \qquad (A5)$$

and Λ for a contact of equal size spheres is

$$\Lambda = R\left(\frac{1 - \delta^2}{E}\right) \qquad (A6)$$

where

P_N normal load, lb

S_{max} maximum Hertz stress, psi

R radius of curvature of sphere, in.

δ Poisson's ratio

E Young's modulus, psi

Since it has been assumed that $Sr_x = Sr_y$, equations (A2) and (A3) are equal. For ease of discussion, only equation (A2) will be referred to.

Substituting equations (A5) and (A6) into equation (A4) results in

$$\tau_{max} = -0.205\left(\frac{3}{2\pi}\frac{P_N}{S_{max}}\right)^{1/2}\left[\frac{E}{R(1 - \delta^2)}\right] \qquad (A7)$$

If equation (A7) is substituted in equation (A2),

$$\left(\tau_{max}\right)_r = -0.205\left(\frac{3}{2\pi}\frac{P_N}{S_{max}}\right)^{1/2}\left[\frac{E}{R(1 - \delta^2)}\right] - \frac{1}{2}\left(\pm Sr_y\right) \qquad (A8)$$

For most steels, $E = 30\times10^6$ psi and $\delta = 0.30$. Therefore, substituting these values in equation (A8) yields

$$\left(\tau_{max}\right)_r = -3.22\times10^6 \left(\frac{P_N}{R^2 S_{max}}\right)^{1/2} - \frac{1}{2}\left(\pm Sr_y\right) \tag{A9}$$

where $\pm Sr_y$ can be either compressive or tensile in nature depending on the sign preceding it. If the residual stress is compressive, the magnitude of the maximum shearing stress is reduced. If, however, the residual stress is tensile, then the magnitude of maximum shearing stress is increased.

APPENDIX B

DERIVATION OF RESIDUAL STRESS EFFECT ON FATIGUE LIFE

Reference 12 states that, for point contact of two rolling elements,

$$L \propto \left(\frac{1}{S_{max}}\right)^9 \tag{B1}$$

where L is life and S_{max} is the maximum Hertz stress within the rolling element contact interface. In reference 11 the following relation is given:

$$\tau_{max} \propto S_{max} \tag{B2}$$

Substituting equation (B2) into equation (B1) results in

$$L \propto \left(\frac{1}{\tau_{max}}\right)^9 \tag{B3}$$

If the residual stresses are present at the point of the maximum shearing stress, the residual stress would increase or decrease this shearing stress as explained in appendix A.

Substituting the change in the maximum shearing stress τ_{max} in equation (B3) yields

$$L \propto \left[\frac{1}{\left(\tau_{max}\right)_r}\right]^9 \tag{B4}$$

where from appendix A

$$\left(\tau_{max}\right)_r = \tau_{max} - \frac{1}{2}\left(\pm Sr_y\right) \tag{B5}$$

Substituting equation (B5) into equation (B3) gives

$$L \propto \left[\frac{1}{\tau_{max} - \frac{1}{2}\left(\pm Sr_y\right)}\right]^9 \tag{B6}$$

Rewriting equation (B4) in a proportional form and assuming a constant Hertz stress result in

$$\frac{L_1}{L_2} = \left[\frac{\tau_{max} - \frac{1}{2}\left(\pm Sr_y\right)_2}{\tau_{max} - \frac{1}{2}\left(\pm Sr_y\right)_1}\right]^9 \tag{B7}$$

REFERENCES

1. Bisson, Edmond E.; and Anderson, William J.: Advanced Bearing Technology. NASA SP-38, 1964.

2. Zaretsky, Erwin V.; Parker, Richard J.; and Anderson, William J.: Effect of Component Differential Hardnesses on Rolling-Contact Fatigue and Load Capacity. NASA TN D-2640, 1964.

3. Bush, J. J.; Grube, W. L., and Robinson; G. H.: Microstructural and Residual Stress Changes in Hardness Steel Due to Rolling-Contact. Trans. ASM, vol. 54, no. 3, Sept. 1961, pp. 390-412, discussion, pp. 818-824.

4. Almen, J. O.: Effects of Residual Stress on Rolling Bodies. Rolling Contact Phenomena, J. B. Bidwell, ed., Elsevier Pub. Co., 1962, pp. 400-424.

5. Scott, R. L.; Kepple, R. K.; and Miller, M. H.: The Effect of Processing-Induced Near-Surface Residual Stress of Ball Bearing Fatigue. Rolling Contact Phenomena, J. B. Bidwell, ed., Elsevier Pub. Co., 1962, pp. 301-316.

6. Carter, Thomas J.; Zaretsky, Erwin V.; and Anderson, William J.: Effect of Hardness and Other Mechanical Properties on Rolling-Contact Fatigue Life of Four High-Temperature Bearing Steels. NASA TN D-270, 1960.

7. Cullity, B. D.: Elements of X-Ray Diffraction. Addison-Wesley Pub. Co., 1956.

8. Jones, A. B.: Metallographic Observations of Ball Bearing Fatigue Phenomena. Symposium on Testing of Bearings, ASTM, 1947, pp. 35-52; discussion, pp. 49-52.

9. Bear, H. Robert; and Butler, Robert H.: Preliminary Metallographic Studies of Ball Fatigue Under Rolling-Contact Conditions. NACA TN 3925, 1957.

10. Carter, T. L.; Butler, R. H.; Bear, H. R.; and Anderson, W. J.: Investigation of Factors Governing Fatigue Life with the Rolling-Contact Fatigue Spin Rig. ASLE Trans., vol. 1, no. 1, Apr. 1958, pp. 23-32.

11. Jones, A. B.: New Departure-Analysis of Stress and Deflections. Vols. I and II, New Departure, Div. Gen. Motors Corp., Bristol (Conn.), 1946.

12. Lundberg, G.; and Palmgren, A.: Dynamic Capacity of Rolling Bearings. Acta Polytech., Mech. Eng. Ser., vol. 1, no. 3, 1947.

NASA-Langley, 1965 E-2789